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SUMMARY

An analog computer simulation was used to study valve-speed-control methods for a turboalternator in a two-spool solar-Brayton system. Steady-state studies of throttling and bypassing were made to determine which locations in the system could be used for the speed-control valves without causing system problems, such as operation of the compressor in the surge region or increased power drain from the heat-storage component. For the selected valve location, dynamic studies were made to determine the response requirements of the valve-actuating equipment. Finally, some of the control equipment design considerations were analyzed, such as the valve sizes and the estimated power requirements of the valve-actuation system.

From the steady-state studies, a throttle valve at the alternator-drive turbine outlet and another valve in a bypass duct around the turbine and throttle valve were selected for the speed control. Variation of valve coefficients with valve movement must provide system flow at or slightly above the design value. If steady-state system flow and, hence, turbocompressor speed are permitted to go below the design value, good response of the turboalternator speed control is unattainable. Prohibition of overdesign power usage from the heat-storage component sets the upper limit on system flow. A $6\frac{1}{2}$-inch butterfly valve for the throttle and a $3\frac{1}{4}$-inch butterfly valve for the bypass keep the system flow at or slightly above the design value. In order to meet the selected speed-control specifications, the valve actuator must have a natural frequency greater than 10 cps. The power loss from throttle-valve pressure drop and bypass-valve leakage is estimated to be 334 watts (about $3\frac{1}{3}$ percent of turbine power). A hydraulic valve-actuation system is estimated to require 260 watts if alternator load varies 1.0 percent continuously and if full-load is switched on and off every 10 minutes.

INTRODUCTION

The Brayton cycle is currently receiving considerable attention for use in space electric power systems capable of producing several kilowatts of power for 1 year or longer. A two-spool solar-Brayton power system is being investigated at the Lewis Research Center (refs. 1, 2, and 3). This system uses a parabolic collector to focus the rays of the sun into a heat-storage component, referred to as an absorber. Sufficient heat is absorbed from the sun to melt a salt, lithium fluoride, in the absorber and to transfer the required heat to the working gas, argon. The melted lithium fluoride supplies its latent heat to the working gas during the shade period of the orbit. The hot gas drives two separate rotating units: a turbocompressor and a turboalternator; thus, it was designated "two-spool." The turboalternator generates approximately 8.75 kilowatts of electrical power. The closed Brayton cycle is completed with a recuperator and a radiator.

The speed of the turboalternator must be controlled closely since electrical frequency is directly proportional to speed, and changes in frequency would affect the operation of some loads. Two general methods of controlling turboalternator speed are commonly used: in the first method, the turbine input power is manipulated with valves to match the turboalternator power to the load power; in the second method, a parasitic electrical load is manipulated to match the alternator load to the power the turboalternator is delivering. Both methods of speed control have advantages and disadvantages, and the relative merits of one method over the other depend on the power system involved and the intended application of the system. A study of a parasitic-load speed control for this system is discussed in reference 4. This report presents the analytical evaluation that was made of the valve method for the two-spool solar-Brayton system.

An analog computer simulation of the Brayton system with its speed control was used to evaluate the valve-speed-control method. The simulation methods are discussed in this report. Steady-state studies were made first to determine the proper location of the valve, or combination of valves, in the gas loop. The criterion used to locate these valves was whether satisfactory control could be achieved for full-power changes without concurrent system problems, such as compressor operation in the surge region or extra power drain from the absorber. For the valve location selected, dynamic studies were made to determine the response requirements of the valve-actuating equipment. Speed-control performance was determined for various actuator responses by stepping alternator load on and off. Finally, the hardware was considered. The required valve characteristics, as well as the power requirements of the actuation system, were analyzed.

SYMBOLS

A_s	surface area, ft^2
C_V	valve flow coefficient
c_v	specific heat at constant volume, ft-lb/(lb mass)-(^{O}R)
E_f	recuperator effectiveness
G(S)	valve actuator transfer function
G_f	specific gravity of argon, based on air
h	heat-transfer coefficient, ft-lb/(ft^2)-(sec)-(^{O}R)
I	polar moment of inertia, (ft-lb)-(sec^2)
K	constant for relating flow to pressure drop
K'	constant relating hA_s/c_v to $w^{0.8}$
K_c	controller gain
K_{cs}	controller gain at which system becomes unstable
L	length, ft
N	speed, rps
P	pressure, lb/ft^2
$\mathscr{P}$	power, ft-lb/sec
Q	torque, ft-lb
R	gas constant of argon, 38.6 ft-lb/(lb mass)-(^{O}R)
S	Laplace operator
T	temperature, ^{O}R
t	time, sec
V	volume, ft^3
v	velocity, ft/sec
W_n	natural frequency, rad/sec
w	mass flow rate, lb mass/sec
Δ	increment
δ	damping ratio
η	alternator efficiency

θ actuator rotation, deg

ρ density, lb/ft^3

τ time constant, sec

Subscripts:

A alternator

B bearing

C compressor

EL electrical load

L load

s sink

T turbine

TA turboalternator

TC turbocompressor

t temperature

v valve

W windage

w flow

1 inlet

2 outlet

SELECTION OF VALVE LOCATION

The speed of a turbine can be controlled by matching the power developed by the turbine to the load on the turbine. In a two-spool Brayton system, the practical way of rapidly changing turbine power is to have a valve, or valves, change the turbine flow rate. A block diagram illustrating the general principle of control is shown in figure 1. The turboalternator speed (or alternator frequency) is sensed and compared with a reference value. The error signal is amplified by the controller and used to operate the valve. The valve changes the turbine flow rate, causing a change in turbine torque in the proper direction to increase or decrease speed dependent on the direction of the error.

In the closed Brayton loop, a number of valve locations allow manipulation of the flow to the alternator-drive turbine. The first step in this study was to determine the location, or locations, which provide the best potential for speed control without introducing other problems into the Brayton system operation. This location was found by

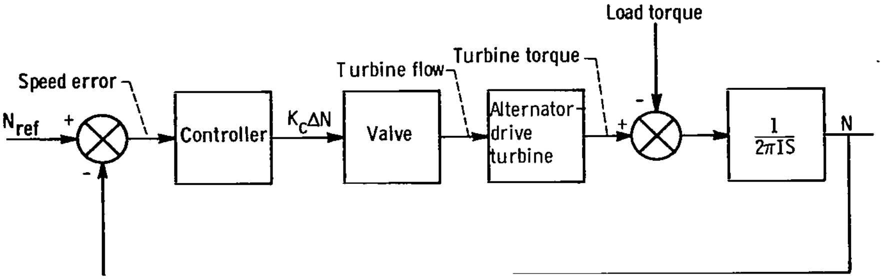


Figure 1. - Block diagram of turboalternator speed-control system.

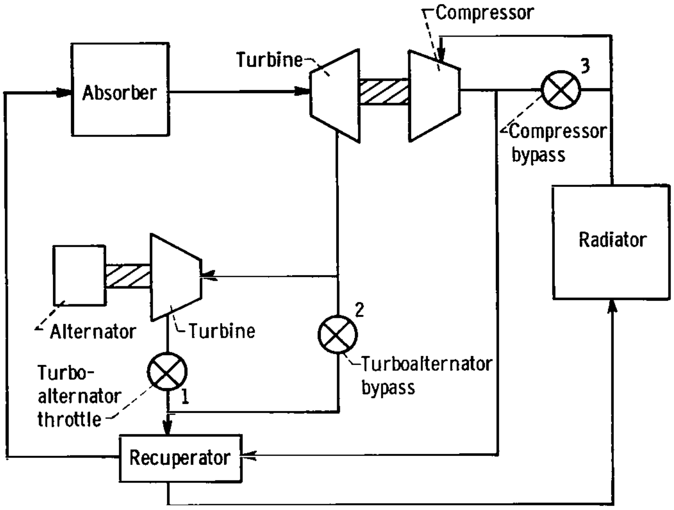


Figure 2. - Valve locations for single-valve methods.

using an analog computer simulation of the Brayton system to map the steady-state effects of system throttling and various bypass schemes. The following valve locations were studied:

(1) A throttle valve at the outlet of the alternator-drive turbine (valve 1, fig. 2)

(2) A valve in a bypass duct around the alternator-drive turbine (valve 2, fig. 2)

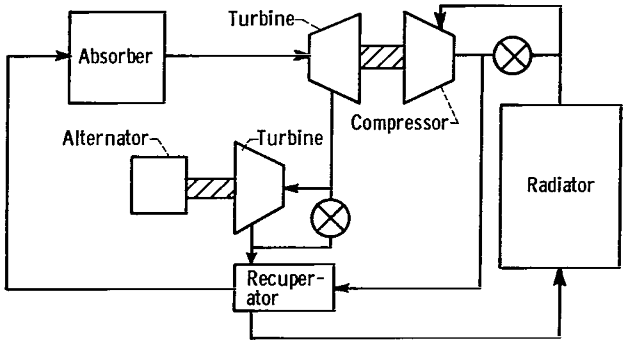
(3) A valve in a bypass duct from compressor outlet to inlet (valve 3, fig. 2)

(4) Two valves, one in a bypass duct around the alternator-drive turbine and the other in a bypass duct from compressor outlet to inlet (fig. 3(a))

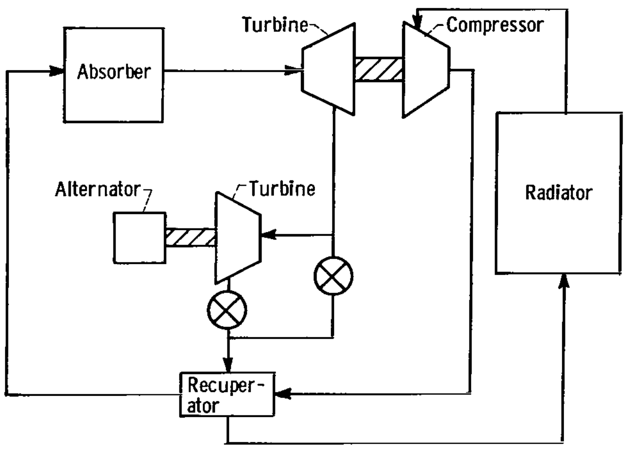
(5) Two valves, a throttle valve at the exit of the alternator-drive turbine and another valve in a bypass duct around the turboalternator and its throttle valve (fig. 3(b))

Computer Simulation

The analog computer simulation of the Brayton system consists of models of all the components integrated into a complete system. A schematic diagram of the simulated system is shown in figure 4. For simulation of the pressure flow dynamics, the system is lumped into five volumes, and each volume is connected by a resistance. Three of



(a) Turboalternator and compressor bypasses.



(b) Turboalternator throttle bypass.

Figure 3. - Valve locations for two-valve methods.

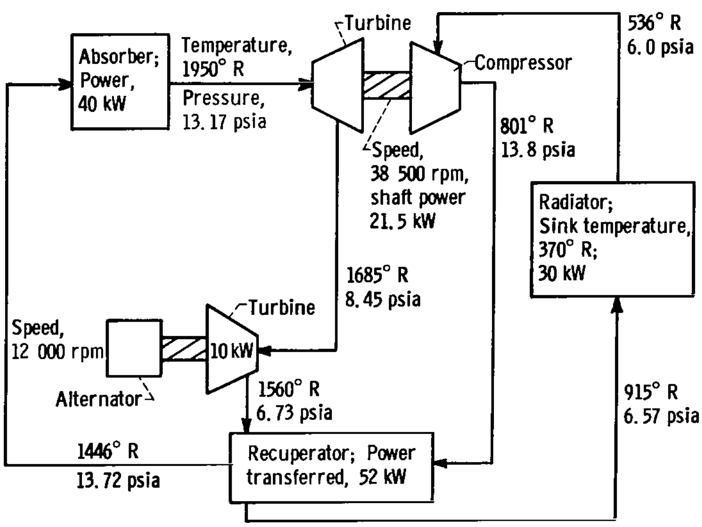


Figure 4. - Two-spool solar-Brayton power system. Argon gas design flow, 0.611 pound per second.

the resistances represent the two turbines and the compressor, while the other two resistances represent the pressure drops of the heat-exchanger equipment and piping. The pressure in each of the volume lumps is computed continuously from the continuity equation:

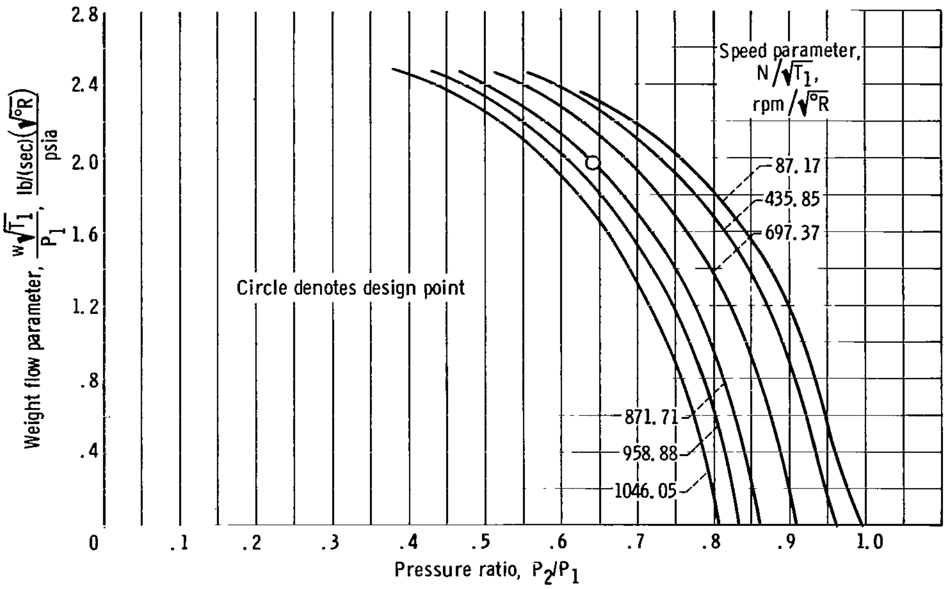
$$w_1 - w_2 = \frac{V}{R} \frac{d\left(\frac{P}{T}\right)}{dt} \tag{1}$$

where w_1 is flow into the volume, and w_2 is flow out of the volume. The flow through the resistances, representing the heat exchangers and piping, was calculated from an orifice-type equation, the assumption being that most of the loss is due to entrance and exit losses in the heat exchangers. The equation for the pressure losses is

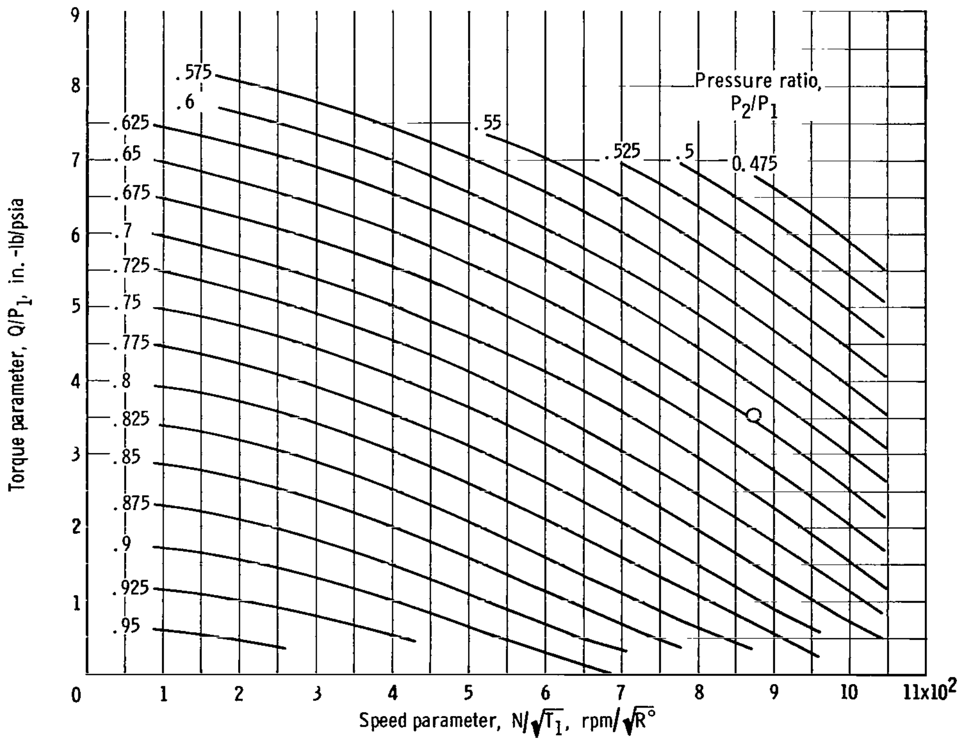
$$w = K(\rho \, \Delta P)^{1/2} \tag{2}$$

where ρ is gas density. The pressure losses in the system are shown in figure 4.

The rotating-machinery simulations are based on three-variable estimated performance maps. For the turbocompressor, two maps were used to simulate the turbine (fig. 5) and two to simulate the compressor (fig. 6). When the pressure ratio and speed are known, the turbine and compressor flow rates

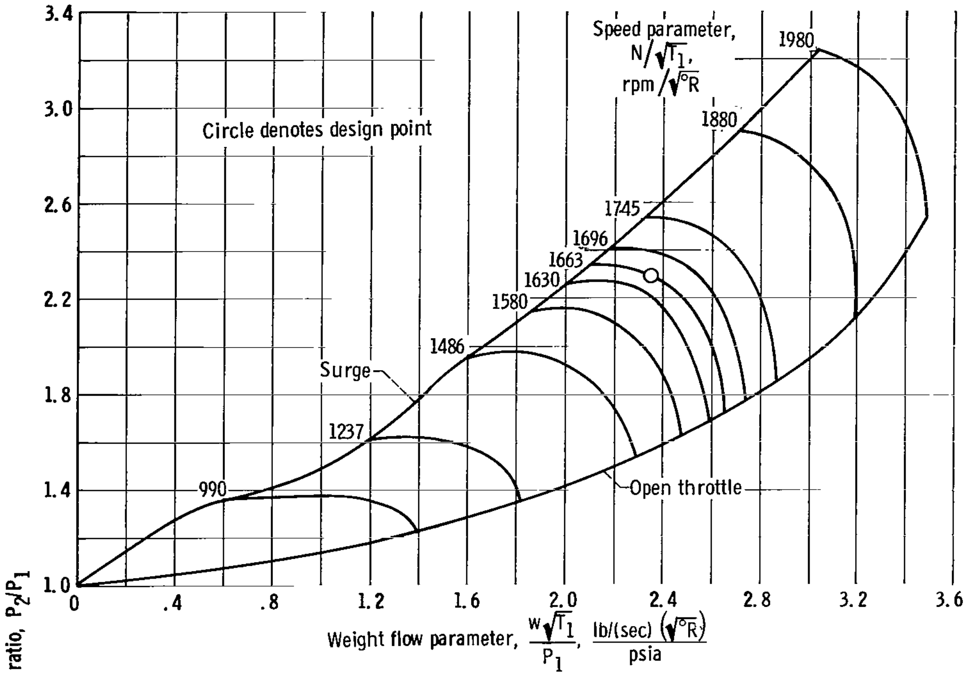


(a) Variation of turbine weight flow parameter with pressure ratio and speed parameter.

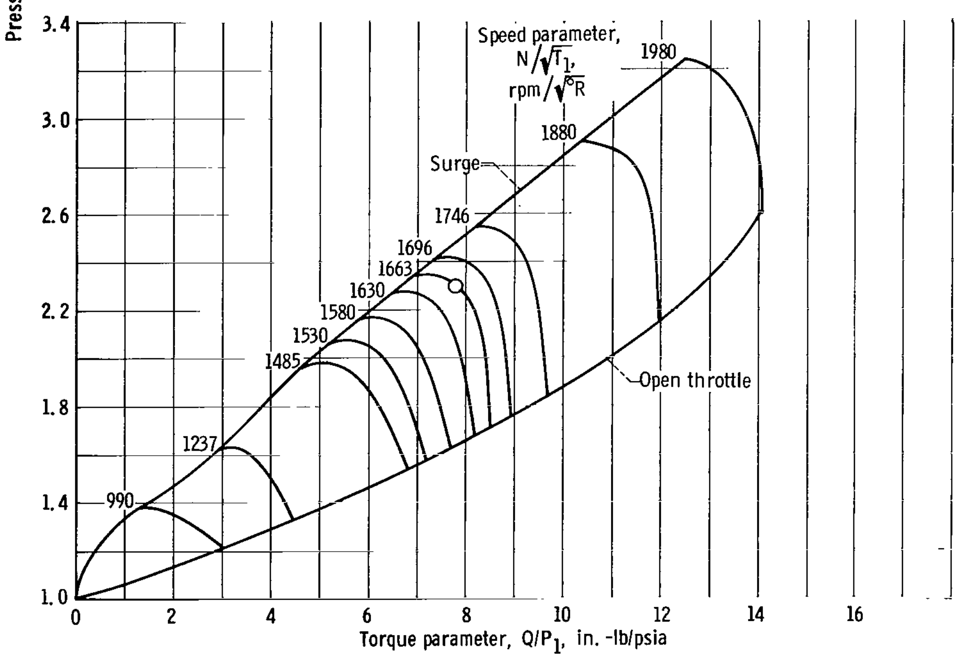


(b) Variation of turbine torque parameter with speed parameter and pressure ratio.

Figure 5. - Estimated compressor-drive turbine maps.

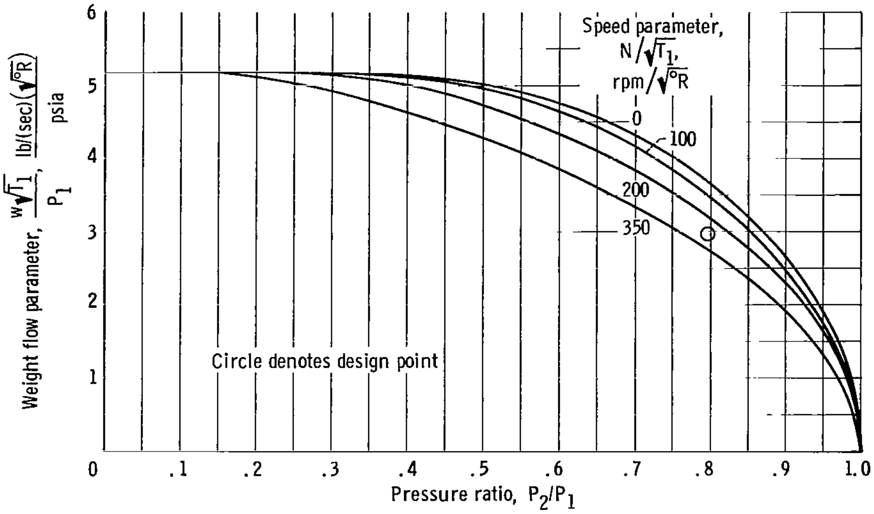


(a) Variation of compressor pressure ratio with weight flow parameter and speed parameter.

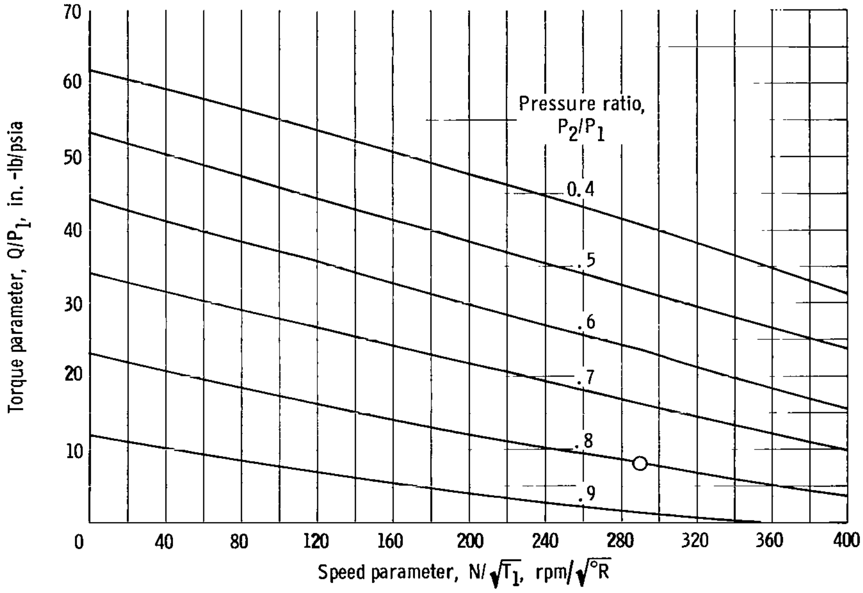


(b) Variation of compressor pressure ratio with torque and speed parameter.

Figure 6. - Estimated compressor performance maps. Impeller design speed, 38 500 rpm.

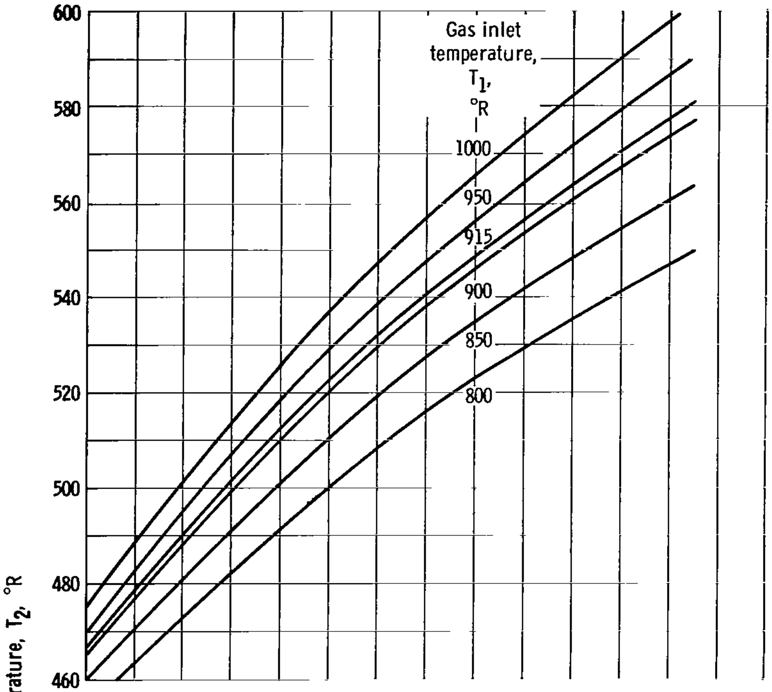


(a) Variation of flow parameter with pressure ratio and speed parameter.

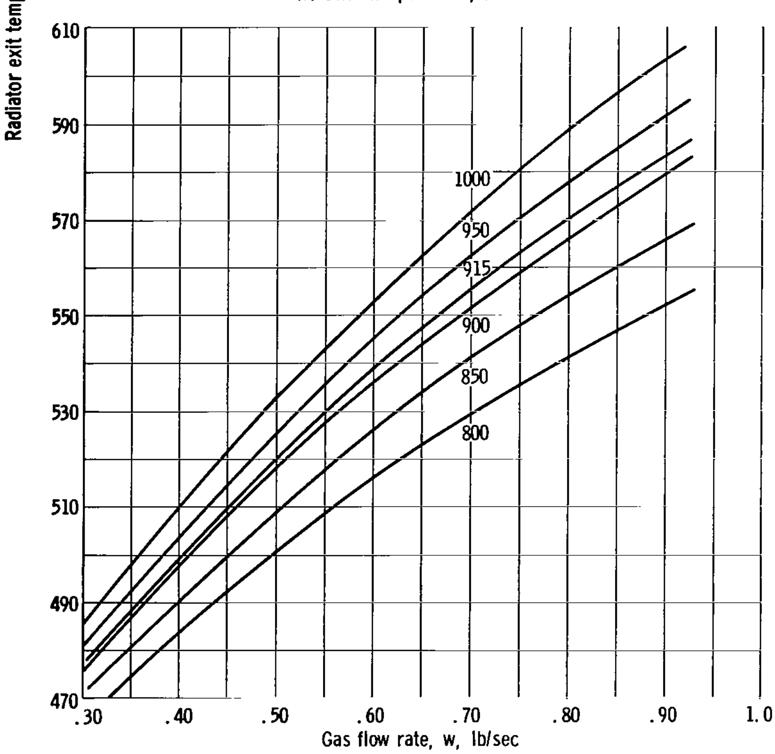


(b) Variation of turbine torque parameter with speed parameter and pressure ratio.

Figure 7. - Estimated alternator-drive turbine performance maps.



(a) Sink temperature, 350° R.



(b) Sink temperature, 400° R.

Figure 8. - Variation of radiator gas exit temperature with gas flow rate and gas inlet temperature.

can be determined from the maps. These flow rates are then used in the appropriate volume lumps to determine the pressure ratio inputs for the map reading. From the combined inertia of the turbocompressor and the torque determinations from the appropriate performance maps, the speed of the turbocompressor is computed continuously (eq. (3)):

$$Q_T - Q_C = 2\pi I_{TC} \frac{dN}{dt} \tag{3}$$

where $I_{TC} = 0.0042$ (ft-lb)-(sec^2).

For the alternator-drive turbine simulation, two three-variable performance maps (fig. 7) were also used. For the steady-state studies, the turboalternator speed was held constant at design speed by programming the alternator torque equal to the turbine torque. The transient determination of turboalternator speed for the dynamic studies is discussed in a later section of the report.

The gas radiator is simulated by using steady-state performance maps. These maps were obtained from a digital computer study made for the Brayton system radiator. The maps show gas exit temperature as a function of gas inlet temperature, gas flow rate, and effective radiator sink temperature. Two of the maps used for the simulation are shown in figure 8. Each map is at a different effective radiator sink temperature.

The radiator dynamics are simulated by transfer functions for outlet temperature to each of the input variables: inlet temperature, flow rate, and sink temperature. The transfer functions were obtained from a separate analog computer study by using a dynamic 10-lump radiator simulation:

$$\frac{T_2(S)}{T_1(S)} = \frac{1}{(\tau_t S + 1)^3} \qquad \text{where } \tau_t = 490 \text{ sec}$$

$$\frac{T_2(S)}{w(S)} = \frac{1}{\tau_w S + 1} \qquad \text{where } \tau_w = 1060 \text{ sec}$$

$$\frac{T_2(S)}{T_s(S)} = \frac{1}{\tau_s S + 1} \qquad \text{where } \tau_s = 650 \text{ sec}$$

In the absorber simulation, it was assumed that the temperature of the lithium fluoride surrounding the gas-flow tube did not vary with position along the tube or with time. Pressure drop along the tube was assumed zero, and the gas density was treated as a constant. With the preceding assumptions, the gas outlet temperature was determined by the following equation:

$$T_{out}(t) = T_{LiF} + \left[T_{in}(t) - T_{LiF}\right]\exp\left(-\frac{L}{v}\,S\right)\exp\left(-\frac{hA_s}{wc_v}\right) \tag{4}$$

In the dead-time term $\exp\left[-(L/v)S\right]$ in equation (4), the time L/v is 0.05 second, and the attenuation term $\exp(-hA_s/wc_v)$ is 0.166 at design flow. Because of the large attenuation of input temperature changes in equation (4), the dead-time term is not very significant and was neglected for the studies discussed herein. In the attenuation term, the heat-transfer coefficient h is assumed a function of Reynolds number to the 0.8 power. This relation is based on the Dittus-Boelter equation. Thus, the attenuation term is $\exp(-K'/w^{0.2})$, where K' is determined from the design conditions. With these additional assumptions, the equation for the gas outlet temperature of the absorber reduces to the following:

$$T_{out}(t) = T_{LiF} + \left[T_{in}(t) - T_{LiF}\right]\exp\left(-\frac{K'}{w^{0.2}}\right) \tag{5}$$

The recuperator is a cross-flow heat exchanger with a design effectiveness of 0.85. For the steady-state studies, the effectiveness equation (eq. (6)) was used to determine both the hot- and cold-gas outlet temperatures (the gas inlet temperatures were known):

$$\frac{\left(T_{in} - T_{out}\right)_{cold}}{\left(T_{in}\right)_{hot} - \left(T_{in}\right)_{cold}} = \frac{\left(T_{in} - T_{out}\right)_{hot}}{\left(T_{in}\right)_{hot} - \left(T_{in}\right)_{cold}} = E_f \tag{6}$$

The effectiveness E_f in equation (6) was related to flow rate (fig. 9). For the dynamic studies, the heat-capacity effects of the recuperator were simulated by programming it in 6 lumps.

For the steady-state studies, the effect of throttling action on the system was simulated by subtracting the desired amount of pressure drop from the pressure at the system location where throttling was being studied. The bypass loop around a system component was simulated by subtracting the desired amount of bypass flow and adding it to the system flow at the inlet and outlet of the bypass loop, respectively. At the location in the system where bypass flow mixes with system flow, a mixed gas temperature was calculated. The valve simulations used for the dynamic studies will be discussed in the DYNAMIC STUDY section.

Results of Valve Location Study

The steady-state studies showed that throttling system flow with a valve at the alternator-drive turbine outlet (fig. 2, p. 5) causes the compressor operating point to enter the surge region when the alternator-drive turbine power has been reduced by only 3.4 kilowatts (figs. 10 and 11). In figure 10, the operating curve for throttling is shown in the compressor performance map, and in figure 11, alternator-drive power and compressor performance parameters are shown as functions of system pressure drop. As shown in figure 10, the throttling curve intersects the surge line at a pressure ratio

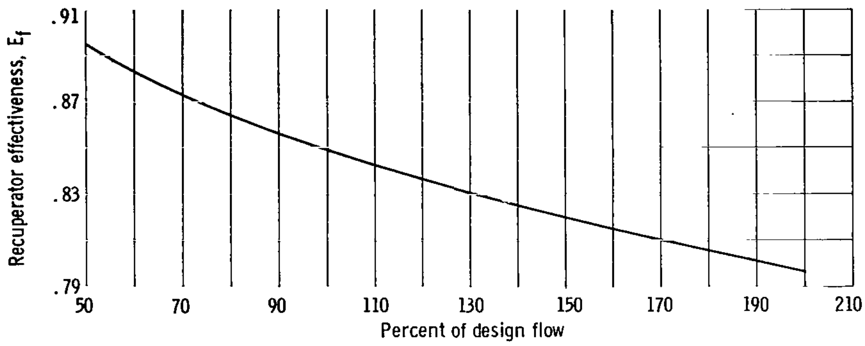


Figure 9. - Variation of recuperator effectiveness with gas flow rate. Design flow, 0.611 pound per second.

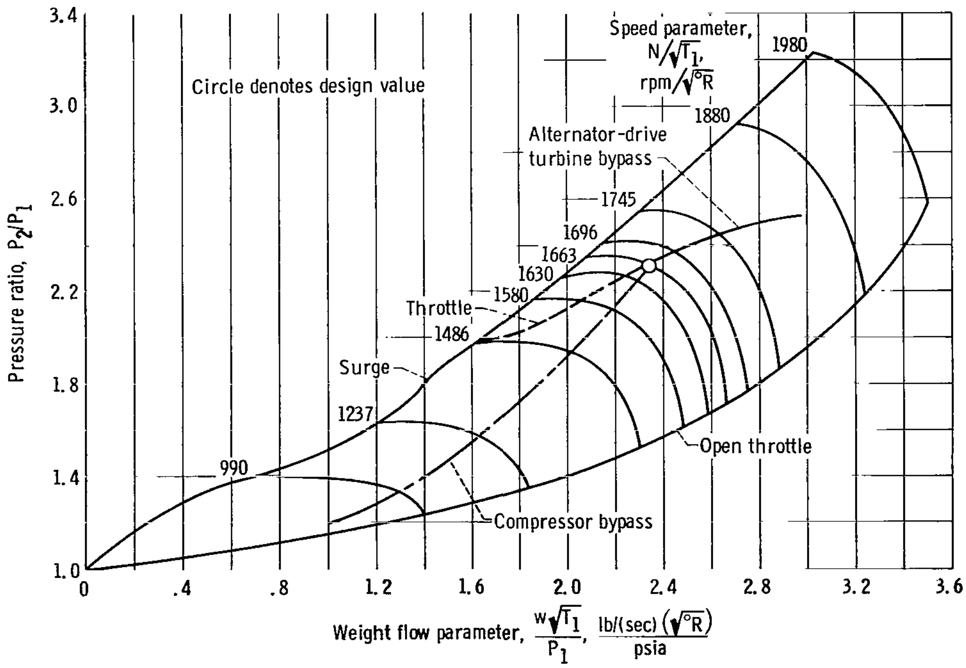
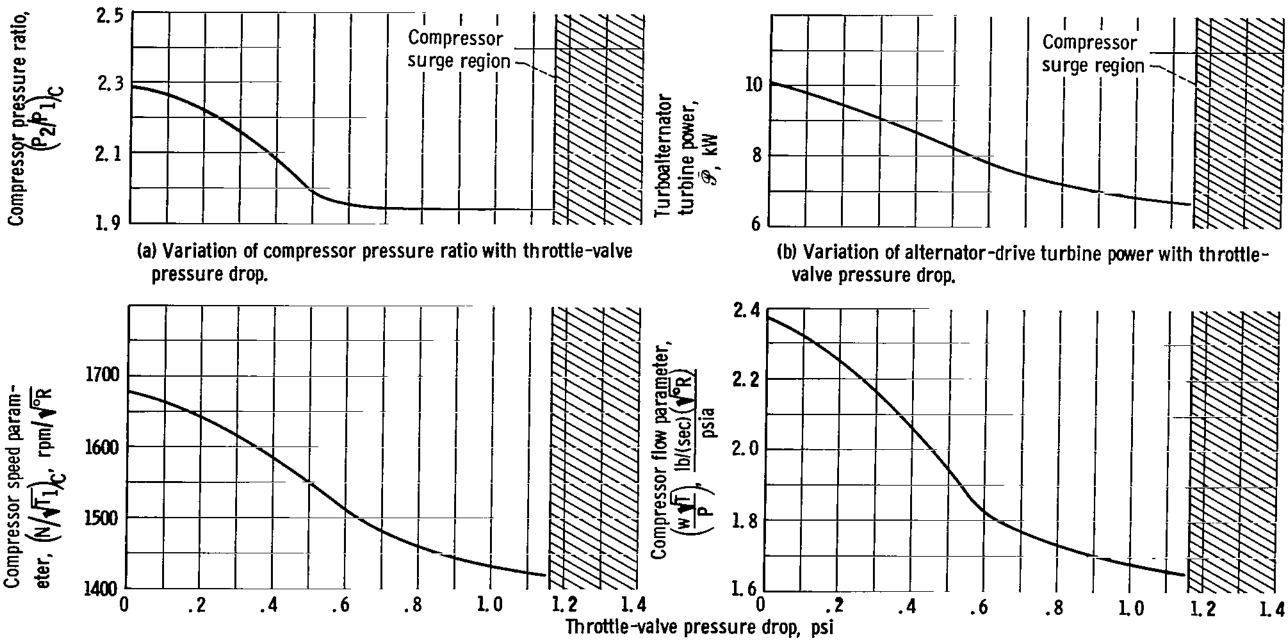


Figure 10. - Compressor operation for system throttling, turboalternator bypass, and compressor bypass. Design speed, 38 500 rpm.



(a) Variation of compressor pressure ratio with throttle-valve pressure drop.

(b) Variation of alternator-drive turbine power with throttle-valve pressure drop.

(c) Variation of compressor speed parameter and flow with throttle-valve pressure drop.

(d) Variation of compressor flow parameter with throttle-valve pressure drop.

Figure 11. - System throttling with valve at turboalternator turbine outlet.

of 1.97, a flow parameter of 1.65, and a speed parameter of 1420. As seen in figure 11, this compressor operating point is reached with a pressure drop of 1.16 pounds per square inch absolute, and the corresponding reduction in turboalternator turbine power is only 3.4 kilowatts from the design value of 10 kilowatts. Therefore, throttling is not considered a satisfactory method of turboalternator speed control since it cannot accommodate full alternator-load changes without introducing the problem of compressor operation in the surge region. The reason for studying the effect of a throttle valve at the turboalternator outlet and not at the inlet is that a close-coupled packaging of the two turbines is advantageous, which makes it difficult to place a valve between the two turbines.

Bypassing flow around the turboalternator turbine (fig. 2, p. 5) reduces turbine power by reducing flow through the turbine, and causes the compressor operating point to move away from the surge region (fig. 10). As flow is bypassed, however, the combined flow resistance of the turboalternator turbine and bypass is less than the design point resistance of the turbine alone; thus, combined flow increases over the design value of the system flow. The increased system flow causes an increase in the power taken from the absorber, which is undesirable since sufficient heat-storage capacity in the absorber for sun-shade missions is a critical aspect of system design. The increases in system flow and absorber power with bypass flow are shown in figure 12 along with the corresponding reduction in alternator-drive turbine power. With this method, a reduc-

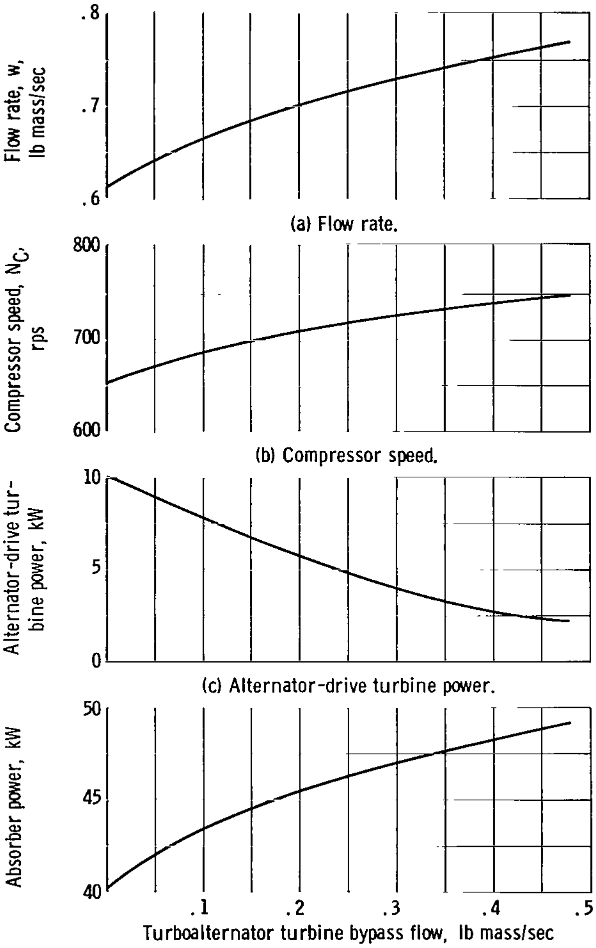
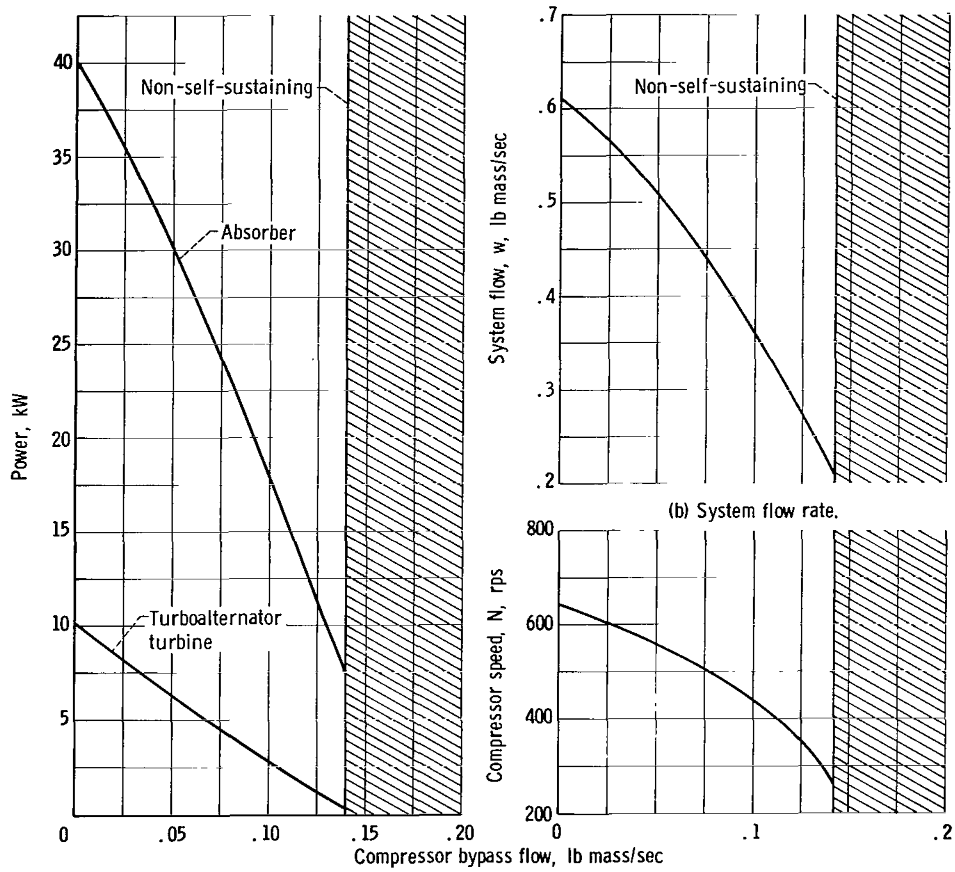


Figure 12. - Variation of system variables with turbo-alternator bypass flow.

tion in alternator-drive turbine power to 2.5 kilowatts causes absorber power to increase to 48.5 kilowatts, which is 21.3 percent over the design value of 40 kilowatts. Also shown in the figure is the variation of compressor speed with bypass flow. The increasing compressor speed, of course, compounds the effect of reduced resistance and explains the high sensitivity of system flow to bypass flow.

Bypassing flow around the compressor from outlet to inlet (fig. 2) also provides a means of reducing alternator-drive turbine flow and power. This method reduces system flow without the attendant reduction in compressor flow that resulted in compressor operation in the surge region, as occurred with the system-throttle valve. Instead, an operating curve nearly parallel to the boundary of the surge region is attained by compressor bypass (fig. 10). Figure 13 shows the variation of absorber power, compressor speed, system flow, and alternator-drive turbine power with compressor bypass flow.

This method, however, is an unacceptable speed control for two reasons. First, an increase in bypass flow sufficient to reduce turboalternator power to near zero also reduces the flow to the compressor-drive turbine. The reduced flow to the compressor-drive turbine results in insufficient torque for the turbine to drive the compressor, and this results in the system becoming non-self-sustaining. Thus, full-load removal could not be accommodated in the system studied with this form of control. The second reason for not considering this form of speed control is based on a decrease in compressor speed with an increase in bypass flow. If the turboalternator were operating at powers below the design value, the resulting decrease in compressor speed would present a problem in dynamic response when full alternator load is applied suddenly. For this condition, turboalternator speed would decrease and could only be corrected by removing some of the load and not reapplying it until the turbocompressor speed and, hence, turboalternator input flow had regained their design values. This fact was verified by



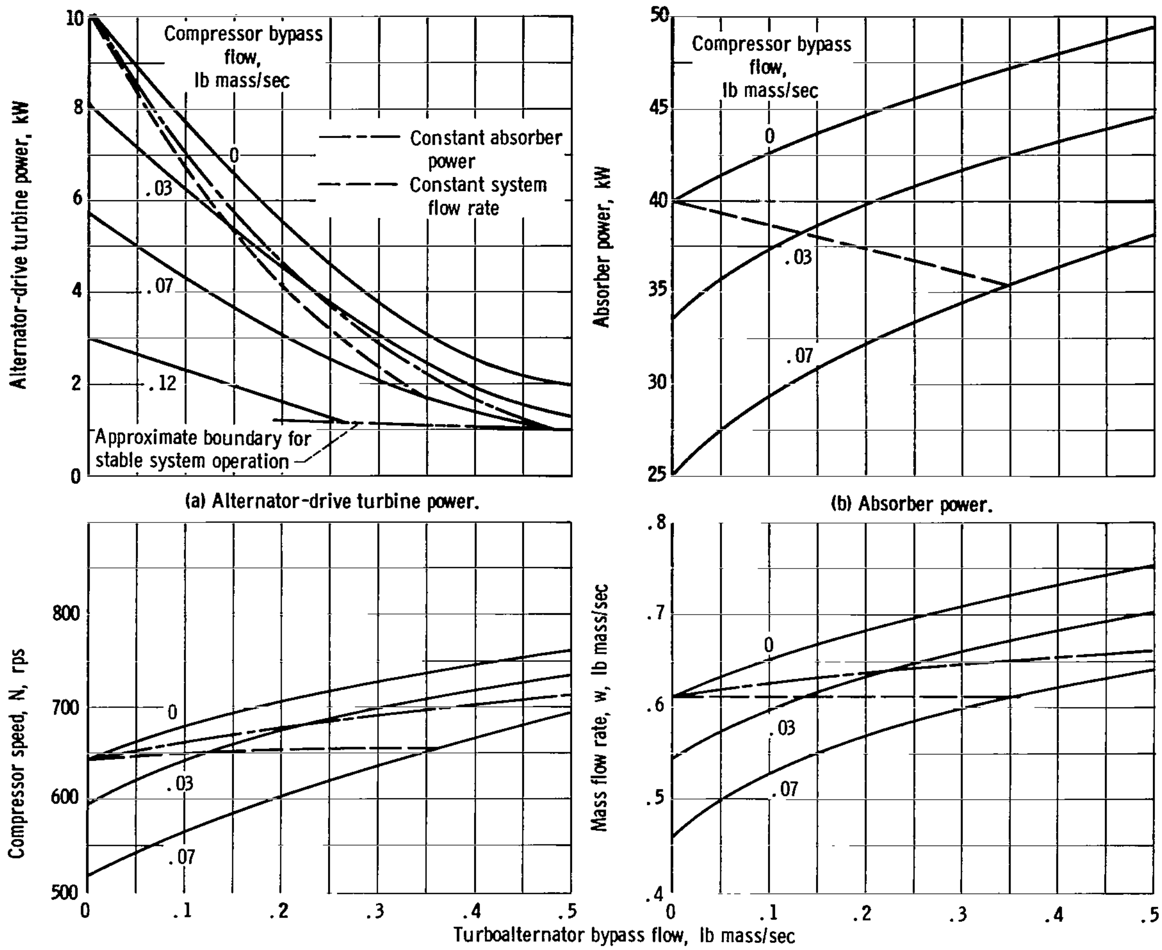
(a) Absorber and turboalternator turbine power. (c) Compressor speed.

Figure 13. - Variation of system variables with compressor bypass flow.

some exploratory runs with a computer simulation of this control method. The computer studies showed that compressor bypass could not control the turboalternator speed with the selected specifications unless the load was added at steps of 1 kilowatt or less.

It should be mentioned that the compressor bypass method has the very attractive advantage of using a single valve in a low-temperature location of the Brayton loop. For a turboalternator with a higher power consumption at the condition of no net electrical output, sufficient compressor bleed might be attained for full-range speed control without causing the system operation to die out. The restriction regarding the rate of load addition could be tolerated perhaps in some applications because of the distinct gain in dependability and packaging offered by this control method.

From the studies discussed thus far, it is apparent that a single valve will not provide a good speed control in the two-spool solar-Brayton system. A single valve that reduces only the flow of the alternator-drive turbine (turboalternator bypass) causes increased system flow and, hence, excessive absorber power drain; while a single valve that reduces system flow along with alternator-drive turbine flow (throttle or compressor



(a) Alternator-drive turbine power.

(b) Absorber power.

(c) Compressor speed.

(d) System flow.

Figure 14. - Variation of alternator-drive turbine power, absorber power, compressor speed, and system flow with turboalternator bypass flow and compressor bypass flow.

bypass) cannot regain either flow quickly because of the reduced speed of the turbocompressor. The throttle valve, of course, was eliminated because of surge region operation before its response potential could be analyzed.

With two valves, however, turboalternator flow can be manipulated while keeping system flow near the design value. The first two-valve scheme studied used one valve in a bypass loop around the turboalternator and the other in a bypass loop from the compressor outlet to the inlet (fig. 3). For this scheme, the variation of alternator-drive turbine power, system flow, turbocompressor speed, and absorber power is plotted in figure 14 as a function of turboalternator bypass flow at different values of compressor bypass flow. The dashed lines in the figure show the combination of the two bypass flows that maintains operation at the design value of system flow and very nearly at the design value of turbocompressor speed. Note that such operation results in a decrease in absorber power. The dot-dash lines in figure 14 indicate operation with slightly increasing system flow, but with constant absorber power. Operating modes between the dash and dot-dash lines would be satisfactory.

This valve combination, however, has the disadvantage of changing the turbocompressor operating point even though the speed is maintained near the design value. Changing the operating point causes the torque required to drive the compressor to become larger than that supplied by the turbine. This increase causes the system to be non-self-sustaining for alternator-drive turbine powers of approximately 1 kilowatt or less (fig. 14). Since the no-load power for the alternator-drive turbine is only 0.52 kilowatts, the two-bypass-valve method was eliminated as a possible means of controlling turboalternator speed. The two-bypass-valve method has the advantage of not introducing valve pressure losses into the system at design operation and could be used to control the speed of a turboalternator having no-load losses greater than 1 kilowatt.

Another two-valve speed control method studied consisted of a throttle valve at the alternator-drive turbine outlet and another valve in a bypass loop around the turbine - throttle-valve combination (fig. 3). This method has the advantage of providing a means for manipulating alternator-drive turbine power while keeping the compressor-drive turbine and compressor at or very near their design operating points. Another advantage of this method is that the throttle valve provides a means for reducing the turboalternator flow to zero, if necessary; thus, only a small bypass valve is needed.

The steady-state maps for this control method (fig. 15) show system flow, compressor speed, absorber power, and turboalternator turbine power as functions of bypass flow and throttle-valve pressure drop. The combinations of bypass flows and throttle-valve pressure drops that provide constant system flow while decreasing alternator-drive turbine power from design to zero are indicated by the dashed lines in the figure. It should be noted that absorber power again decreases slightly with decreasing turboalternator power. The dot-dash lines in figure 15 show the combinations of bypass flow and pressure drop needed to maintain design absorber power with slightly increasing system flow. For system operation between the dashed and dot-dash lines, absorber power requirements will be at or below the design value, and, in addition, the dynamic response of the system should be satisfactory because it is not necessary for the turbocompressor to accelerate during alternator load additions.

From these steady-state studies, therefore, it is concluded that the combination of throttling and bypassing of the alternator-drive turbine provides the best means of controlling turboalternator speed with valves in this Brayton system. Proper control of the two valves keeps the power system near its design operation point even for large variations in alternator power requirements. The control criterion for the valves is to maintain system flow at or slightly above their design values. The upper flow limit is set by the design value of absorber power. The lower flow limit (design flow) is set by the

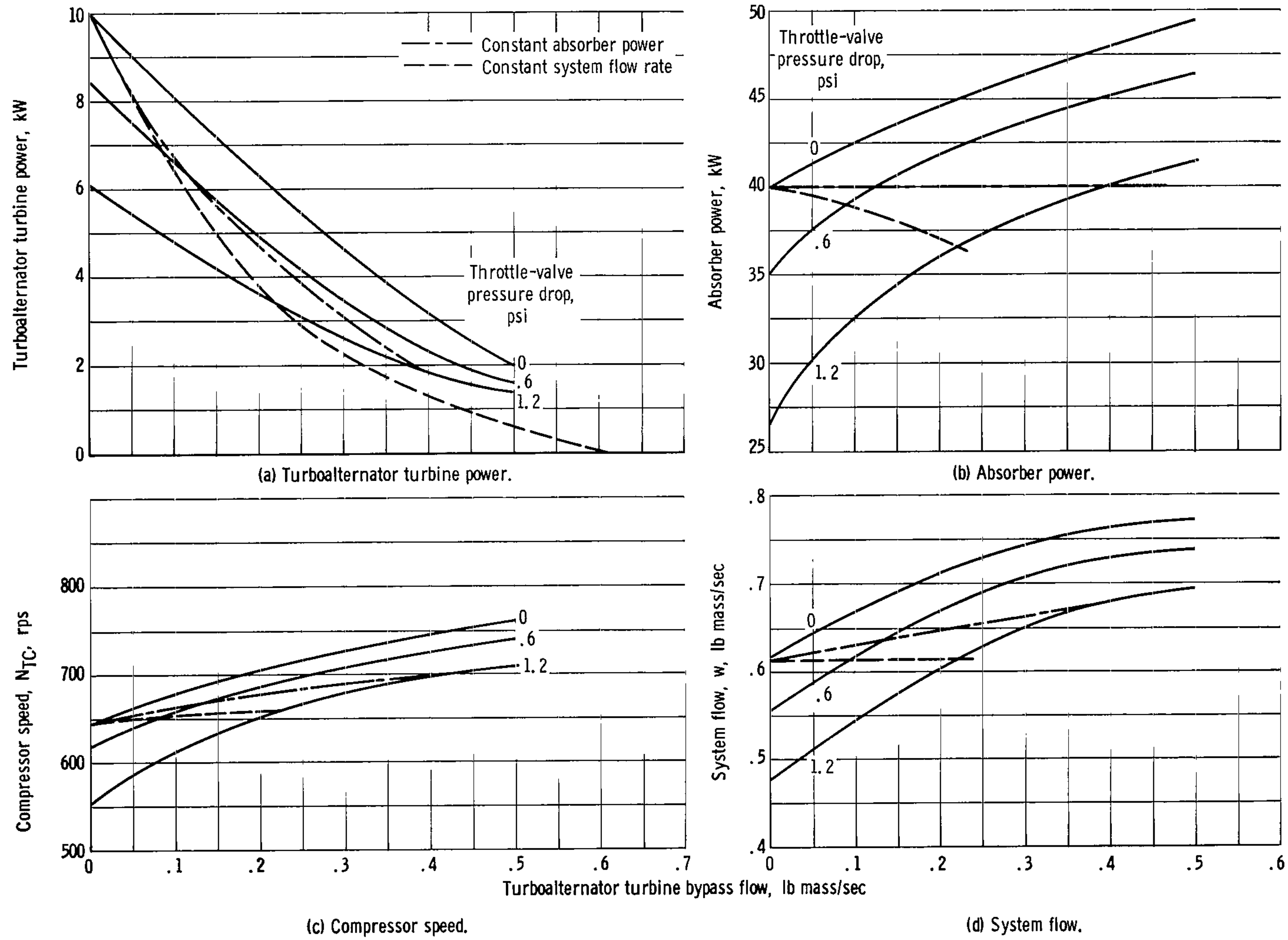


Figure 15. - Variation of alternator-drive turbine power, absorber power, compressor speed, and system flow with turboalternator turbine bypass flow and exit throttle-valve pressure drop.

requirement of design compressor speed.

DYNAMICS STUDY

After the turboalternator throttle-bypass method of speed control was selected, dynamic studies were made to determine the response requirements of the control. In order to make the dynamic studies, the dynamics of the control components and the turboalternator had to be included in the computer simulation.

Computer Simulation

A block diagram of the dynamic simulation of the speed-control system is shown in figure 16. The first block of the control system is the controller. For these studies, a proportional controller was used with the gain set to reduce the turbine torque to the no-load value at the specified value of steady-state speed error. No-load torque is the torque required to overcome bearing and alternator windage losses.

The valve dynamics are represented in the second block of figure 16. For this study, it was assumed that both valves were operated from one actuator. The overall valve transfer function G_s of the degree of rotation of the valve shafts θ divided by amplified speed error $K_c \Delta N$ is

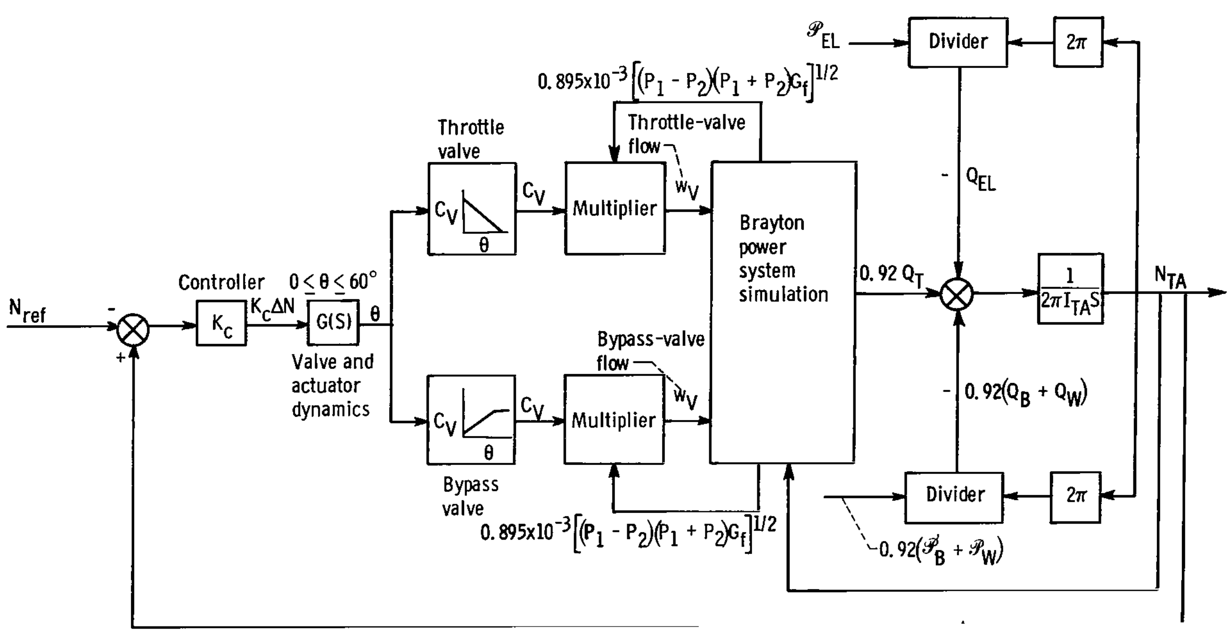


Figure 16. - Block diagram of dynamic simulation of turboalternator throttle-bypass speed-control system.

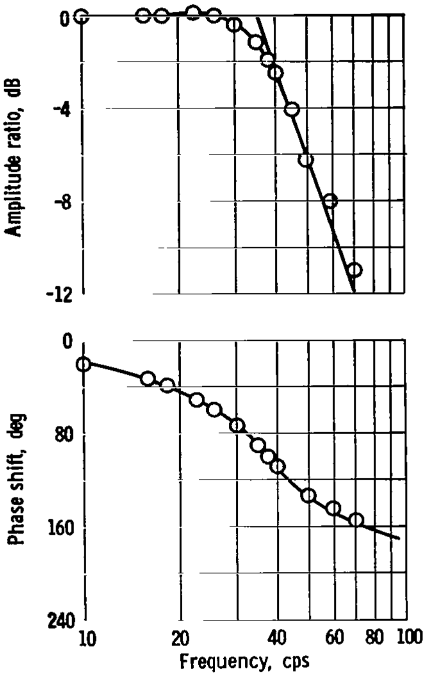


Figure 17. - Variation of amplitude ratio and phase shift from output to input with frequency for 4-inch hydraulically operated butterfly valve.

$$G(S) = \frac{\theta}{K_c \, \Delta N} = \frac{1}{\left(\frac{S}{W_n}\right)^2 + 2\delta\left(\frac{S}{W_n}\right) + 1} \tag{7}$$

where W_n is the natural frequency and δ is the damping ratio. This transfer function was used in this study because data for hydraulically actuated butterfly valves, which are typical for this application, exhibit a second-order characteristic. For example, the data shown in figure 17 are for a 4-inch hydraulically actuated butterfly valve tested at the Lewis Research Center. The corner frequency (natural frequency) is 35 cps, and the asymptote to the amplitude locus has a slope of -12 decibels per octave, which is indicative of a second-order system. The phase shift is 90° at the corner frequency of 35 cps and approaches 180° at higher frequencies, which is also a characteristic of second-order systems.

Valve manufacturers have adopted a valve flow coefficient for rating control valves. This coefficient C_V is defined as

$$C_V = \frac{q}{\left(\frac{P_1 - P_2}{G}\right)^{1/2}} \tag{8}$$

where q is the volumetric flow rate through the valve in gallons per minute, P_1 minus P_2 is the valve pressure drop in pounds per square inch, and G is the specific gravity of the flowing fluid, based on water. Stated another way, C_V for a given valve is equal to the gallons per minute of water flowing when the valve is subjected to a pressure drop of 1 pound per square inch. For gas flow, it is convenient to express equation (8) in terms of mass flow rate, density, and a gas specific gravity G_f. The gas specific gravity G_f is based on a specific gravity of air equal to 1 at a pressure of 14.7 pounds per square inch absolute and a temperature of 520° R. The gas density is approximated by an average value by using the upstream and downstream density and the perfect gas law. For gas flow, the valve coefficient C_V is defined as

$$C_V = \frac{1118\, w_V}{\left[(P_1 - P_2)(P_1 + P_2)G_f\right]^{1/2}} \tag{9}$$

where w_V is the mass flow rate through the valve and G_f is the specific gravity of the fluid (at the flowing temperature), based on air.

For this study, the valve coefficient C_V for the throttle and bypass valves was assumed to vary linearly over the 60° range of shaft rotation θ, as indicated in figure 16. At the Brayton system design point, the throttle valve was open 60° at a C_V of 1300. This value of C_V is typical of an 8-inch butterfly valve open 60°. The bypass valve was fully closed at the design point, and when this valve was open 60°, a C_V of 210 was used. This value of C_V corresponds to that of a $3\frac{1}{4}$-inch butterfly valve open 60°. This valve size results in a pressure drop across the bypass valve approximately equal to the alternator-drive turbine pressure drop at design; thus, a system flow is assured that is close to the design value when at the no-load condition. From the values of C_V for the two valves and the pressure information in the Brayton simulation discussed previously, the throttle and bypass flows were computed from equation (9) in the manner indicated in figure 16.

With the turbine flow determined and with the turbine maps included in the Brayton simulation discussed previously (fig. 7), sufficient information was available to compute the responses of turbine torque and speed to variations in load power. This was done with the use of the equations for the inertial dynamics of the turboalternator:

$$Q_T - Q_L = 2\pi I_{TA} \frac{dN}{dt} \tag{10}$$

The coefficient I_{TA} is the moment of inertia of the turboalternator, and for this study it was equal to 0.051 (ft-lb)-(sec^2). The load torque Q_L in equation (10) is the sum of the turboalternator bearing power $\mathscr{P}_B$, alternator windage loss $\mathscr{P}_W$, power loss in the alternator $\mathscr{P}_A$, and the electrical load power $\mathscr{P}_{EL}$ divided by the speed:

$$Q_L = \frac{\mathscr{P}_B + \mathscr{P}_W + \mathscr{P}_A + \mathscr{P}_{EL}}{2\pi N} \tag{11}$$

For the gas bearings, a design power loss of 300 watts was assumed. The bearing power loss is proportional to the speed squared, and for small speed changes the percentage change in power is approximately equal to twice the percentage change of speed. For speed changes of 2 percent, this means only a change of 4 percent in bearing power, or a 0.12-percent change in the turbine power (12 W out of 10 000 W). Because the bearing power changes so little for the speed errors expected in the dynamic studies, the bearing power was programmed as a constant. A design point power loss of 220 watts was assumed for the alternator windage loss. Windage loss varies as a function of the speed cubed, but it was also programmed as a constant because speed errors of only 2 percent were expected in the dynamic studies. The alternator efficiency η was considered to be 0.92 and was assumed to be constant for the small speed errors of interest. Therefore, the electrical power loss in the alternator $\mathscr{P}_A$ is given by

$$\mathscr{P}_A = (\mathscr{P}_T - \mathscr{P}_B - \mathscr{P}_W)(1 - \eta) \tag{12}$$

Substituting equation (12) into equation (11) results in the following expression for the total load torque on the turbine:

$$Q_L = (1 - \eta)Q_T + \frac{\eta(\mathscr{P}_B + \mathscr{P}_W) + \mathscr{P}_{EL}}{2\pi N} \tag{13}$$

Combining equation (13) with equation (10) gives the equation programmed on the computer for the turboalternator dynamics:

$$\eta Q_T - \frac{\eta(\mathscr{P}_B + \mathscr{P}_W) + \mathscr{P}_{EL}}{2\pi N} = 2\pi I_{TA} \frac{dN}{dt} \tag{14}$$

The flow of computer information in solving equation (14) is indicated in figure 16.

The electrical load power $\mathscr{P}_{EL}$ in equation (14) was the disturbance function for the dynamic studies of the speed control, being switched off and on in increments up to the design value (8.72 kW). It was assumed that this power did not vary with speed. This assumption implies that the load does not change for small speed changes and that the voltage regulator maintains voltage close to the design-point value during speed transients.

Results of Dynamic Study

In the dynamic speed control studies, several values of valve system natural frequency were used to determine the response needed for satisfactory control. The damping ratio in the valve system transfer function was kept at a constant value of 0.5 for all the studies. Generally, lower values of damping ratio result in a more oscillatory control, while higher values result in a slower control.

The following speed control specifications were used in this study to evaluate the

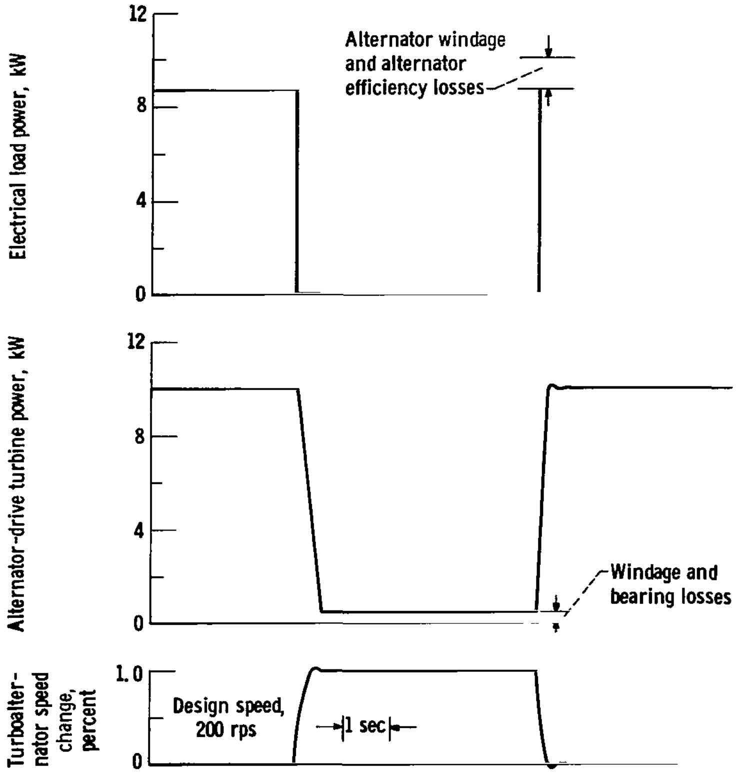


Figure 18. - Response of turboalternator throttle-bypass control. Valve system natural frequency, 30 cps.

speed control performance:

(1) Steady-state speed error of ±1 percent

(2) Maximum transient speed error of 2 percent

(3) Recovery time of 1 second or less to the specified steady-state speed error

(4) No sustained oscillations (i.e., oscillations lasting longer than 10 sec with no decrease in amplitude)

These specifications are to be met for all load changes up to and including full load (8.72 kW).

During operation with reduced alternator load, there is increased heat rejection from the Brayton loop. Unless additional radiating area is provided for this condition, such as fins on the bypass duct, the compressor inlet temperature will slowly rise above its design value. This rise in temperature will result in a reduction in available turboalternator power. This reduction would have to be considered in determining a turboalternator full-load rating consistent with system response capability. In the computer studies, however, the load was switched off and on in a few seconds, and compressor inlet temperature did not have time to increase significantly above design. Therefore, the speed control could handle the full-load rating based on design conditions (8.72 kW).

The first studies were made for full-load switching with a valve system natural frequency of 30 cps and a gain set to reduce the torque from the turbine to the no-load value for 1-percent overspeed in steady state. The results of this test are given in figure 18, which shows alternator-drive turbine power and speed response as load power is switched off and on. Note that the speed specifications just discussed are met and that very little overshoot occurs. The relative stability of the control with the 30-cps natural frequency was determined by increasing the gain until the system became unstable. A gain increase of 50 percent over the value for a 1-percent speed error resulted in small sustained oscillations in speed and flow when full-load power was switched off. The control, therefore, has a gain margin of 0.33 when operating with the gain required for a 1-percent steady-state speed error. Gain margin is defined as follows:

$$\text{Gain margin} = \frac{K_{cs} - K_c}{K_{cs}} \tag{15}$$

where K_{cs} is the gain that will cause instability and K_c is the actual gain. Studies were also made with reduced gain, and the variation of gain margin with steady-state speed error for the 30-cps valve system is shown in figure 19.

Valve systems with natural frequencies less than 30 cps were studied, and for natural frequencies down to 10 cps, the control met the response specifications. In fact, for nautral frequencies of 10 cps or greater, the valve responsed fast enough compared with the turboalternator speed response to limit the transient speed error to about 1 percent when full-load power was switched off. The time required for the turboalternator to reach 1-percent speed error can be calculated from equation (14) by assuming that load torque is zero; for this turboalternator the time was 0.1 second. It was assumed that the valves must operate fully in 0.1 second to keep the speed error at 1 percent or less. It was also assumed that full operation requires a time equal to the time constant of the valves multiplied by four. These assumptions require that the natural frequency of the valve systems must be at least 12 cps if the alternator speed error is to be controlled to less than 1 percent. This calculation was made for a second-order valve system with 0.5 damping. This estimate agrees fairly well with the results obtained from the computer studies. These calculations were made to provide an estimate of the turboalternator speed response and to show how this response affects the response requirements of the control.

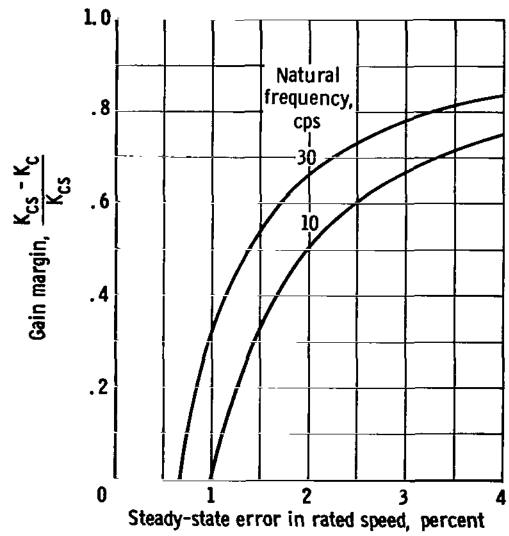


Figure 19. - Variation of gain margin with steady-state speed error for valve systems with natural frequencies of 10 and 30 cps.

Small oscillations in speed and flow were encountered for full-load switching with the natural frequency of the valve system reduced to 10 cps, and the gain set for a 1-percent steady-state speed error. The 10-cps valve system, threfore, does not meet all the speed control specifications and by definition has a gain margin of zero. Increasing the gain results in larger sustained oscillations, while reducing the gain results in larger steady-state speed errors. The variation of gain margin with steady-state error for the 10-cps valve system is also shown in figure 19. Note that increasing the steady-state speed error to 1.5 percent results in a gain margin of 0.33 for the 10-cps system, which is the same as the gain margin for a 30-cps system with a 1-percent speed error.

The limiting factor, therefore, in determining the natural frequency required to meet the speed control specifications was not the transient error but the stability of the control system. A natural frequency greater than 10 cps is required for a stable control system with 1-percent steady-state speed error. Obviously, increasing the allowable steady-state speed error specification will allow the use of a valve system with a lower natural frequency.

HARDWARE REQUIREMENTS

If one actuator is used to control both the throttle and the bypass valves, the valves must have particular C_V characteristics for a particular mode of system operation, such as constant system flow or constant absorber power. As discussed previously, the speed control must function in a manner that maintains system flow at or above the design value but below the overdesign value that increases absorber power output.

Valves

From the steady-state maps for this control (fig. 15), it is possible to determine what C_V contours the valves should have to provide constant system flow or a flow consistent with constant absorber power as the alternator-drive turbine power is reduced. For constant system flow, the bypass-valve C_V contour is calculated by realizing that the pressure drop across the valve will remain nearly constant at the design value of the alternator-drive turbine pressure drop. The C_V contour is then a function of the bypass flow only and can be calculated from equation (16):

$$C_V = \frac{1118\ w_V}{\left[(P_1 - P_2)(P_1 + P_2)G_f\right]^{1/2}} \tag{16}$$

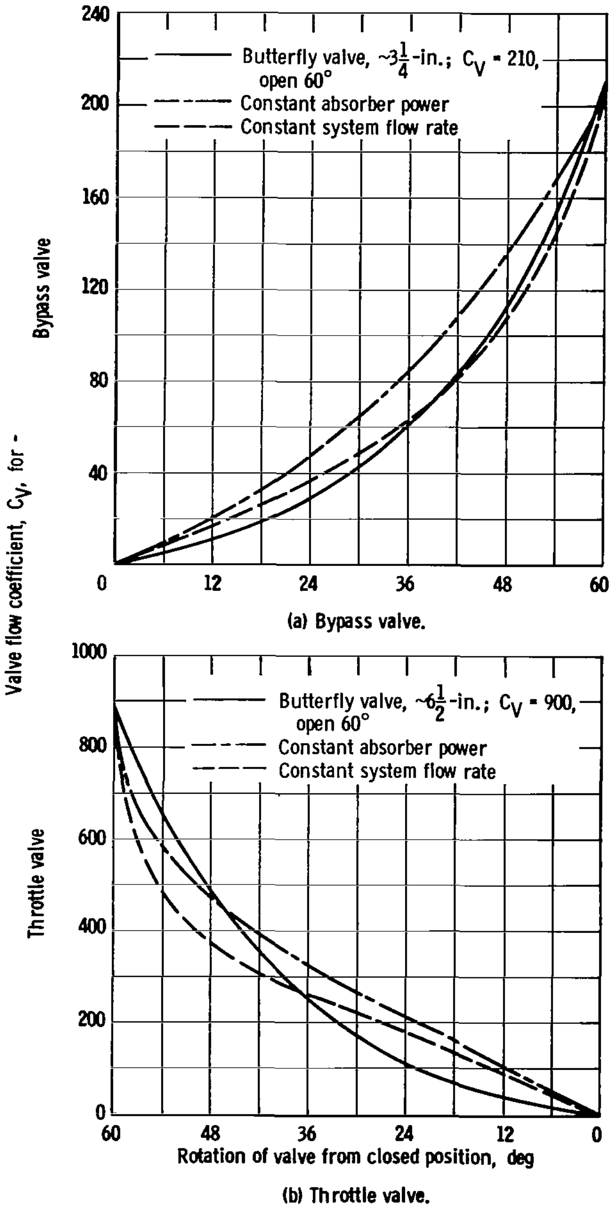


Figure 20. - Valve flow coefficient contours for throttle-bypass speed control.

For this calculation, P_1 and P_2 are upstream and downstream pressures (psia) of the bypass valve, respectively, and G_f is the specific gravity of argon (at the flowing temperature), based on air. The required bypass-valve C_V for constant system flow is shown in figure 20(a).

The required throttle-valve C_V for constant system flow operation is also calculated by using equation (16). The throttle-valve exit pressure P_2 is nearly constant, and P_1 is P_2 plus the throttle-valve pressure drop. This pressure drop is obtained from figure 15 (the intersection of the constant flow lines and the pressure drop lines). The required throttle-valve C_V characteristic for constant system flow operation is shown in figure 20(b).

The values of C_V for the throttle valve and bypass valve for flows consistent with constant absorber power can also be calculated by using equation (16) and the maps of figure 15; however, because system flow changes, the system pressures change and the values obtained from the computer studies must be used for the calculations. These calculated C_V values for the throttle and bypass valves are also shown in figure 20.

If the valves have C_V values that lie between the curves of figure 20(a) and (b) for constant flow and constant absorber power, the turboalternator turbine power can be reduced from design to zero power without decreasing system flow below the design value or increasing absorber power above the design value. Such valves, therefore, satisfy the requirements, as previously discussed. Also shown in figure 20 are the C_V curves for a $3\frac{1}{4}$-inch butterfly valve for bypassing and a $6\frac{1}{2}$-inch butterfly valve for throttling; these are both for 60° operation. Both the $3\frac{1}{4}$- and $6\frac{1}{2}$-inch C_V curves approximate the required bypass and throttle C_V curves fairly well.

As mentioned earlier in the report, a throttle-bypass control will reduce system efficiency because of the throttle-valve pressure drop and the bypass-valve leakage at design operation. The $6\frac{1}{2}$-inch throttle valve at an opening of 60° is estimated to have a design pressure drop of 0.09 pound per square inch, which will cause a power loss of 308 watts. The power loss due to leakage flow through the bypass valve at the design value is estimated to be 26 watts, resulting in a combined power loss of 334 watts for the throttle-bypass scheme at design operation. This loss does not include the actuator power to drive the valves; the actuator power requirements are discussed in the next section.

The foregoing C_V requirements are dictated by the use of a single actuator in the speed control. If two actuators with separate control signals are used, such as turboalternator speed for the throttle valve and system flow or compressor speed for the bypass valve, then the C_V contours of the valves are not as important. Of course, two actuators with separate control signals represent a more complicated speed-control system.

Actuator System

The results of the dynamic studies showed that the valve system must have a natural frequency of 10 cps or greater to control the turboalternator speed to the selected specifications for full-load switching. This response requirement can be met with hydraulic operation. In order to estimate the power needed by the hydraulic system, two requirements of the Brayton power system must be known or estimated: (1) the frequency of full-load cycling of the system load and (2) the amount of system load that is continuously varied.

For the first requirement, an accumulator system can be used to supply the needed power. The accumulator would be sized to provide the quantity of hydraulic fluid needed to move the valve one full stroke times the number of successive full-load switches that may occur during full-load cycling. Some calculations were made to determine the power needed for accumulator charging as a funttion of the time between periods of full-load cycling. These calculations were based on hydraulic tests at the Lewis Research Center on an 8-inch butterfly valve driven by a rotary actuator. In these calculations, the assumption was made that the actuator power as a function of the full-stroke response characteristic of the 8-inch valve reasonably approximates the characteristics of two smaller valves operating from a single actuator. Figure 21 shows the power required by the hydraulic system (motor power to operate hydraulic pump) against the time be-

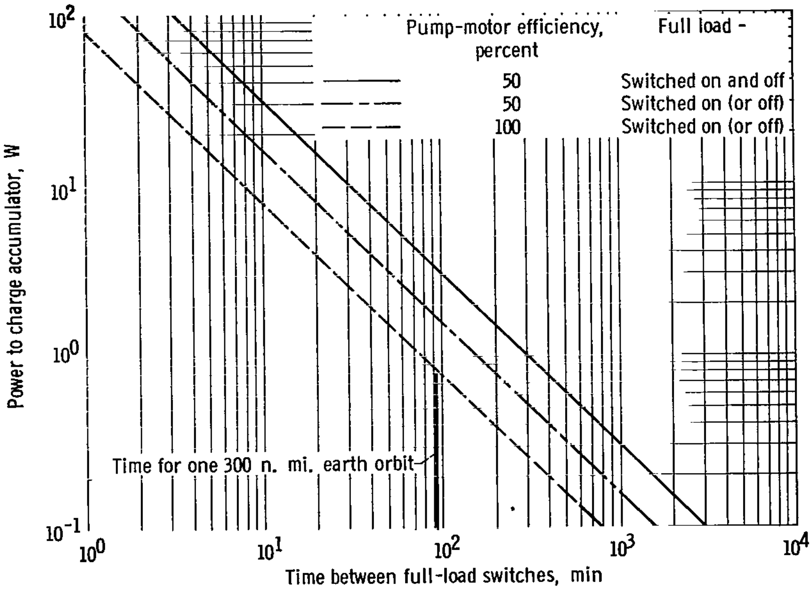


Figure 21. - Variation of power to charge accumulator with time between full-load switches. Hydraulic-system pressure, 3000 pounds per square inch absolute.

tween periods of full-load switching. The dashed line shows continuous power needed to charge the accumulator against the time between periods of full-load switching. This line is for a pump motor with an efficiency of 100 percent and the capacity to pump enough hydraulic fluid into the accumulator for only one full-load switch at a time. The dot-dash curve is for a pump-motor efficiency of 50 percent but still for only one full-load switch. The solid curve is for two successive full-load switches, one immediately after the other, and a pump-motor efficiency of 50 percent. These curves show that the amount of continuous accumulator charging power necessary to handle full-load switching is quite small if periods of switching occur at intervals of 10 minutes or longer. The curve indicates that only 30 watts are required continuously to handle a full-load cycle (two full-load switches) every 10 minutes for a pump-motor efficiency of 50 percent. The size of the accumulator is small, only 14 cubic inches, and provides a margin of safety of 10. The size of the accumulator does not change with the interval of time between periods of load switching.

The second requirement is concerned with the amount of continuous variation of the turboalternator load. This requirement places a higher power requirement on the hydraulic system. Figure 22 shows the estimated power to drive the pump-motor as a function of the continuous load variation. This is shown for pump-motor efficiencies of 50 and 100 percent. The results presented in figure 22 apply to load step changes occurring 0.1 second apart. This is approximately the time required for the valves to operate. The 0.1-second time interval results in the maximum continuous power required. Less hydraulic power is required for longer time intervals between step changes in load. This is accomplished by using a smaller pump with an accumulator.

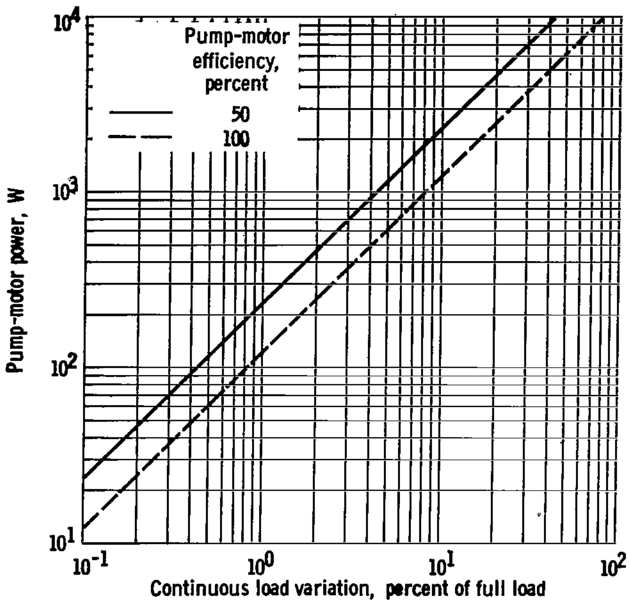


Figure 22. - Variation of hydraulic-system power requirements with continuous load variation. Hydraulic-system pressure, 3000 pounds per square inch absolute. Load switched on and off at 0.1-second intervals.

As shown in figure 22, the hydraulic power requirements vary considerably with the amount of load variation occurring continuously. For a pump-motor efficiency of 50 percent, the continuous power requirements are 230 watts for a 1-percent load variation, but 2300 watts for a 10-percent load variation.

The total hydraulic power requirement is obtained by adding the accumulator-charging power and the power for continuous load variation. The sum is only 260 watts for a full-load cycle every 10 minutes and a 1-percent continuous load variation occurring at 0.1-second intervals. If the continuous variation is 10 percent, however, the sum becomes 2330 watts. It is evident that the continuous duty of the control must be known to predict accurately the power consumption of the valve actuation system.

The total power drain of the speed control from the useful output of the Brayton system is obtained by adding the previously mentioned valve losses of 334 watts to the total hydraulic loss. Again, the total is very much dependent on the continuous requirements of the control. For 1-percent continuous duty, the total calculated power loss to the speed control is 594 watts.

SUMMARY OF RESULTS

A study to evaluate valve schemes for controlling the speed of a turboalternator in a two-spool solar-Brayton power system yielded the following results:

1. Two valves are necessary to control turboalternator speed if the selected response specifications for full-alternator-load changes are to be met and if the system problem of increased absorber power output is to be presented. A throttle valve should be located at the turboalternator turbine outlet and another valve in a bypass loop around the turbine and throttle valve. With this control there is increased heat rejection from the Brayton loop with reduced alternator load. Unless additional radiator area is provided, such as fins on the bypass duct, there will be a reduction in available turboalternator power. This reduction would have to be considered in determining a turboalternator full-load rating consistent with system response capability.

2. In order to meet the transient specifications, the actuator controlling the valves must provide a valve system natural frequency greater than 10 cps.

3. With one actuator for both valves, the valves must have particular valve flow coefficient C_V characteristics. These C_V characteristics must maintain system flow at or above design but below the values resulting in increased absorber power usage for the full range of turbine power. Flows below design cause poor speed control response, and absorber powers above design are critical to system operation. For the Brayton system studied, a $3\frac{1}{4}$-inch butterfly valve in the bypass line and a $6\frac{1}{2}$-inch butterfly valve at the turbine outlet provide the approximate C_V characteristics.

4. The total loss in Brayton system electrical power output due to the speed control depends strongly on the continuous variation expected in electrical load during a mission. If the continuous variation is only 1 percent of the design load, the calculated total loss is 594 watts. This value includes 230 watts for hydraulic pump operation for the continuous duty, 30 watts for hydraulic pumping to a small accumulator that can handle full-load cycles 10 minutes apart, and 334 watts attributed to valve pressure drop and leakage.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, August 17, 1966,
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